Appendix C

Sample Computation

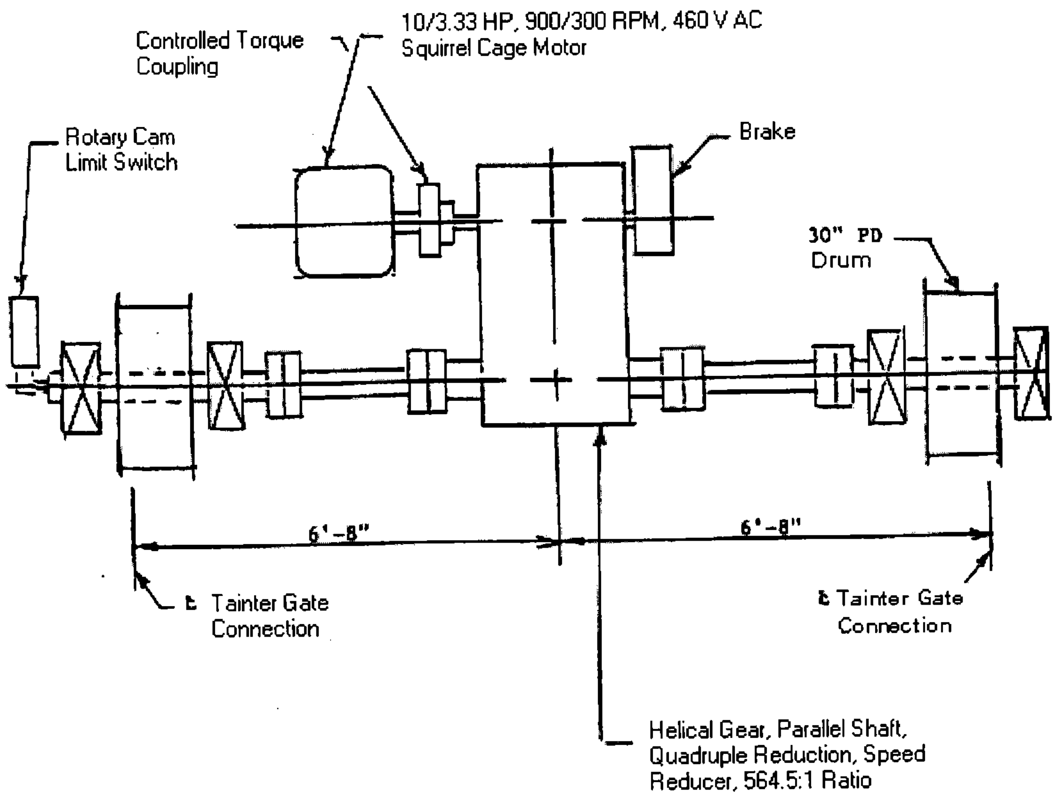
The following sample problems are provided to show methodology used to determine gate hoist operating loads, and component size and strength. The examples show one approach, other approaches and supporting references may also be appropriate.

1. Tainter gate - Electric Motor Wire Rope Hoist. This example provides a design analysis for a wire rope type tainter gate hoist. It shows the relationship among the various load conditions that the hoist may experience during gate operation, both normal and extreme. The detailed design analysis for the hoist is most often a requirement of a performance specification.

2. Sector Gate Machinery Design. This example provides typical calculations for determining a sector gate's closing pintle torque for a reverse head. Closing torque is composed of hydrodynamic forces acting on the nose of the gate, and hinge and pintle friction that is composed of friction from gate's weight and hydrostatic head.

3. Round Link Chain - Grooved Drum and Pocket Wheel. This example provides a detailed design analysis for a round link chain grooved drum and pocket wheel tainter gate hoist. The detailed design analysis for the hoist is most often a requirement of a performance specification. This example provides good information about what a designer should expect when reviewing design submittals.

1. Tainter gate - Electric Motor Wire Rope Hoist. The following is an example of a wire rope hoist design for a small tainter gate.



Design Criteria

Design load - Criteria to determine tainter gate machinery loads can be found in EM 1110-2-2702. For this example use 18,300 lbs for rated hoist capacity.

Assume design load to be split 70/30* between hoist drums

Hoist speed of drive to be as follows:

High speed (hs) = 11.02 fpm

Low speed (ls) = 3.67 fpm

Max P (pull per drum) = 18,300 x .7 = 12,810 lbs.

(normal operating condition)

Note: Torque requirements are the same for low speed and high speed operation; therefore, the design criteria will be based on the torque and horsepower requirements for the high speed operating mode.

* The 70/30 split is a conservative approach for hoist design and is offered as a recommendation based on practical experience.

Required Motor HP

$HP = WV/(33,000 \times E)$

where: W = 18,300 lbs.

V = hoist speed fpm

E = efficiency = 0.90

$HP_{hs} = 6.79$

$HP_{ls} = 2.26$

Use 10/3.33 HP, 900/300 RPM 460V, 3 PHASE, 60 CYCLE
high slip (12% $\pm$ 1%), high torque, squirrel cage constant torque motor

Operating speed @ 12% slip
High Speed = 900 x .88 = 792 RPM
Low Speed = 300 x .88 = 264 RPM

Brake

Brake to be rated at 150% of normal full load motor torque

Normal Torque = T_n = (HP x 63,025)/RPM = 10 x 63,025/792 = 796 lb-in = 66 lb-ft
Rated Brake Torque = T_b = 66 x 1.5 = 100 lb-ft

Use 10 in. diameter wheel rated 150 lb-ft for continuous duty and set to operate at 100 lb-ft.

Motor Coupling

Motor coupling to be controlled torque type with automatic reset. Coupling to be sized to slip at 200% of full load. Torque is based on a maximum motor torque of 325% of normal full load motor torque. Service Factor = 1.0.

Slip Torque = T_s = (HP x 63,025 x service factor x 200%)/RPM
T_s = 10 x 63,025 x 1.0 x 2.0/792 = 1592 lb-in.

Motor shaft size dictates coupling selection.
2 7/8 in. motor shaft diameter
2 3/8 in. reducer shaft diameter

Slip torque range based on ± 20% of setting
High = 1274 lb-in
Mean = 1592 lb-in
Low = 1910 lb-in

Speed Reducer

Reducer Ratio

Based on a 12% motor slip, hoist speed of 11.02 fpm, and a 30 in pitch diameter drum.

Drum RPM (High Speed) = $(11.02 \times 12)/(30 \times \pi) = 1.4031$

Drum RPM (hs/ls) = 1.4031/0.4677
Motor RPM (hs/ls) = 792/264

Ratio = 792/1.403 = 564.5 to 1

Reducer Efficiency = 94%

Overhung Load = 875 lb. (1/2 the weight of the floating shaft assembly)

Reducer Ratings

Min Durability = 1.5 x Motor Rating (1.0 x Brake Rating)
Input hp = 1.5 x 10 = 15
Output torque = 796 in-lb x 1.5 x 564.5 x 0.94 = 633,572 in-lb

Min Strength = 1.5 x Durability Rating
Input hp = 1.5 x 15 = 22.5
Output torque = 633,572 in-lb x 1.5 = 950,358 in-lb

Min Required Overload Rating – Based on motor stall torque
Input hp = 3.25 x 10 = 32.5
Output torque = 796 in-lb x 3.25 x 564.5 x 0.94 = 1,372,740 in-lb

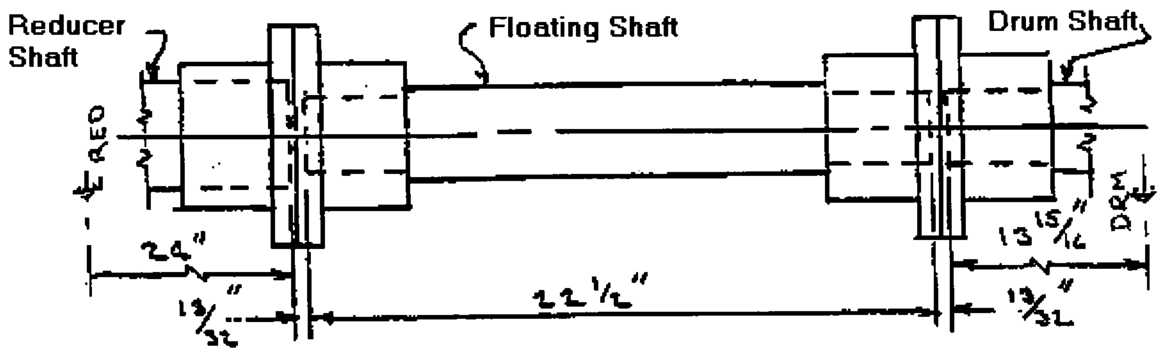
B-10 Bearing Life Requirements

4 cycles/hour and 3 min/cycle
4 x 3 = 12 min/hr
50 year life, operate 9 months/year

Required Life = 12/60 x 24 x 365 x 50 x 0.75 = 65,700 hrs
Use 75,000 B-10 hours

Reducer to be quadruple reduction, parallel shaft, 564.5:1 ratio and will have low speed torque ratings equal to or greater than the high speed torque ratings.

Floating Shaft.



T_n (70/30 split) = 0.94 x 564.5 x 796 x 0.7 = 295,600 lb-in

T_b (brake 50/50 split) = 0.94 x 564.5 x 1200 x 0.5 = 318,400 lb-in

T_m (motor stall torque 70/30 split) = 0.94 x 564.5 x 796 x 3.25 x 0.7 = 960,900 lb-in

Use AISI C-4150 heat treated ASTM-A434 Grade B,C
Tensile Strength (S_{ts}) = 103,000 psi

Yield Point = 80,000 psi (Minimum Yield Point at ½ shaft radius)

Per ASME Code
Allowable Shear Stress = 0.3 x Yield Point or 0.18 x Tensile Strength
Use the lesser of the two
Keyway Factor = 0.75 x Shear Stress

Minimum design stress = Ultimate Strength/5

ASME Code is more conservative and will be used.
103,000 x 0.18 = 18,540 psi Allowable shear stress
18,540 x 0.75 = 13,905 psi Keyway factor, use 10,000 psi

Required Shaft Diameter - Based on Torsional Deflection

$d = 0.1 * \sqrt[3]{T}$ Machinery Handbook 21st Edition, Page 455
(based on 1 degree of deflection in a length 20 times shaft diameter)

$$d = 0.1 * \sqrt[3]{318{,}400} = 6.83 \text{ in., where } T = T_b$$

Required Shaft Diameter - Based on Allowable Shear Stress

$$d = B * \sqrt[3]{\frac{5.1 K_T T}{S_s}}$$ Machinery Handbook 21st Edition, Page 459

B = 1 for solid shafts, K_T = 1.25 for loads suddenly applied with minor shock

$$d = \sqrt[3]{\frac{5.1 x 1.25 x 318{,}400}{10{,}000}} = 5.88 \text{ in., where } T = T_b$$

Torsional Deflection Dictates Shaft Size

Shear Stress Based on Brake Torque (T_b)

6.75 in. Shaft Diameter at Couplings

$S_s = 5.1 \text{ x } K_T \text{ x } T_b / d^3 = 5.1 \text{ x } 1.25 \text{ x } 318{,}400/(6.75)^3 = 6600$ psi

Shear Stress Based on Motor Stall Torque (T_m)

6.75 in. Shaft Diameter at Couplings

$S_s = 5.1 \text{ x } K_T \text{ x } T_m / d^3 = 5.1 \text{ x } 1.25 \text{ x } 960{,}900/(6.75)^3 = 19{,}900$ psi

Endurance Limit for Shaft

$S_e = k_a k_b k_c k_d k_e k_f S_e'$

S_e = Corrected Endurance Limit

k_a = Surface Factor (0.75* Machined)

k_b = Size Factor (0.75*)

k_c = Reliability Factor (0.87)

$k_c = 1 - 0.08\, D_o$ where D_o = Deviation Factor = 1.6 for 95% Survival Rate

k_d = Temperature Factor (1.0*)

k_e = Modifying Factor for Stress Concentration (0.71, 0.75)

$k_e = 1/K_f$, $K_f = 1 + q\,(K_t - 1)$

Shaft Diameter D = 7 in., at Couplings d = 6.75 in., D/d = 1.04

Fillet Radius, r = 0.125 in., r/d = 0.019

From Mechanical Engineering Design by Shigley Chart K_t » 1.4

Use q (maximum) = 1.0, then $K_f = K_t = 1.4$ and $k_e = 1/1.4 = 0.71$

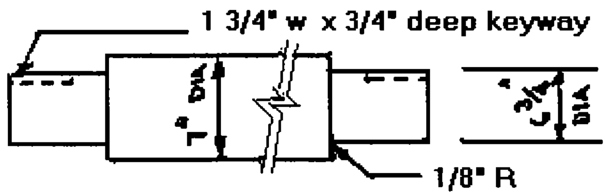
For Keyway, $k_e = 0.75$

k_f = Miscellaneous Effects Factor (0.85 assumed)

$S_e' = 0.5 \text{ x } S_{ts} = 0.5 \text{ x } 103{,}000 = 51{,}500$ psi

* From Mechanical Engineering Design by Shigley, 2nd ed, McGraw Hill

S_e (at Fillet) = 0.75 x 0.75 x 0.87 x 1 x 0.71 x 0.85 x 51,500=15,200 psi
S_e (at Keyway) = 0.75 x 0.75 x 0.87 x 1 x 0.75 x 0.85 x 51,500=16,100 psi
Endurance Limits are greater than brake torque shear stress of 6600 psi. Therefore, based on ASME Code allowable shear stress and endurance limits, shaft size is acceptable.



Couplings – Final Drive

Rigid coupling half to be mounted on the floating shaft ends, flexible coupling half to be mounted on the reducer shaft and the drum shaft. Couplings should have a minimum strength rating of 2 times the vendor's published catalog rating. Couplings should be selected based on a strength rating of 2 and a maximum motor starting torque of 325% x the normal full load motor torque.

T_m (max motor torque 70/30 split) = 960,900 in-lb
T_b (brake torque 50/50 split) = 318,400 in-lb

Change in Length (Reducer to Drum) Due to Thermal Expansion at 140°

$\Delta L = \epsilon t l$

where: ϵ = coefficient of expansion for 100° = 0.00065
t = Temperature in °F
l = Shaft Length in inches

$$\Delta L = \frac{0.00065 x 140 x 61.25}{100} = 0.06 \text{ in.}$$

Use Flex - Rigid Gear Type Couplings for Floating Shaft

	Catalog Rating in-lb	Strength Rating in-lb	Maximum Bore Flex-Sq Key - in.	S.F. Catalog Rating – Based on Brake Torque	S.F. Strength Rating – Based on Max Motor Torque
A	535,500	1,071,000	7.75	1.68	1.11
B	693,000	1,386,000	8.75	2.18	1.44

Use Coupling A

Wire Rope

Try 1" diameter 6x37 IWRC, Type 304 stainless steel, breaking strength = 77,300 pounds. Due to the difference between the drum windings of each hoist drive one drum will have right lay rope and the other drum will have left lay rope. Each pair of wire ropes will be furnished in matched pre-stretched pairs. The drum pitch diameter is 30 times the rope diameter = 30".

Safety Factors

Minimum Safety Factor = 5 (Based on Normal Load)
Minimum Safety Factor = 3 (Based on Peak Load)
Minimum Safety Factor = 1.5 (Based on Motor Stall Torque)

Normal Load

SF = 77,300/(18,300 x 0.5) = 8.45 (50/50)
SF = 77,300/(18,300 x 0.7) = 6.03 (70/30)

Peak Load (Full Load Motor Torque (T_n) at 10 hp)

T_n = 796 in-lb, R = PD/2 = 30/2 = 15", Efficiency = 92%
P(peak load) = (796 x 564.5 x 0.92)/15 = 27,560 lb
SF = 77,300/(27,560 x 0.5) = 5.61 (50/50)
SF = 77,300/(27,560 x 0.7) = 4.0 (70/30)

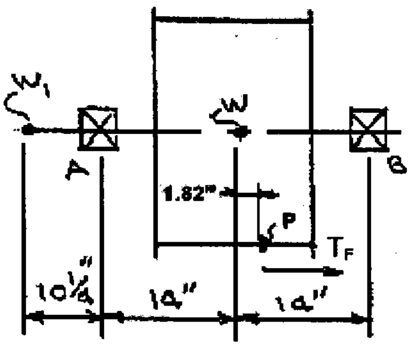
Motor Stall

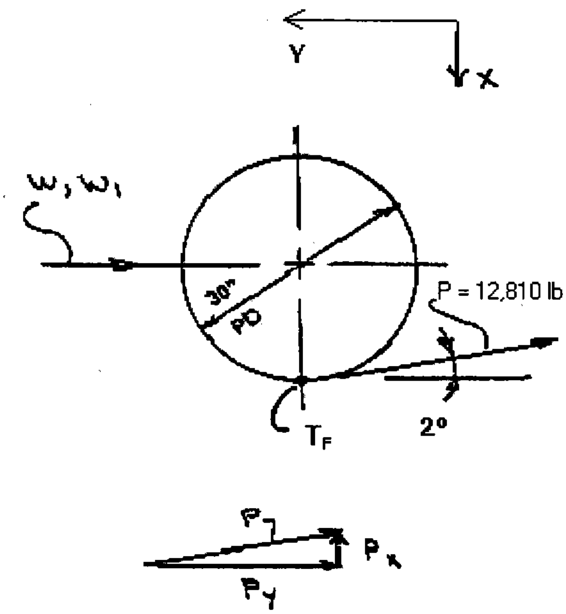
P(motor stall) = (796 x 564.5 x 0.92 x 3.25)/15 = 89,569 lb
SF = 77,300/(89,569 x 0.5) = 1.73 (50/50)
SF = 77,300/(89,569 x 0.7) = 1.23* (70/30)
* Slightly under the minimum but is allowed since this condition will be experienced infrequently, if at all, and because of the other safety devices such as the slip coupling.

Use 1" 6x37 type 304 wire rope.

Hoist Drum Assembly - Normal Operating Condition (Gate Closed)

P (Cable Pull) = 18,300 lb
For 70/30 Split, P = 12,810 lb
W = 2,700 lb (drum assembly)
W_1 = 875 lb (1/2 of floating shaft assembly)
1° = Fleet Angle
2° = Angle of Wire Rope from Drum to Gate
1.82" = Wire Rope ¢ at gate closed position





$P_x = P \sin 2^\circ = 447$ lb
$P_y = P \cos 2^\circ = 12{,}802$ lb
$T_F = P \sin 1^\circ = 224$ lb
$T_{FAX} = - T_{FBX} = (224 \times 15)/28 = 120$ lb

For Stuck Gate (Maximum Motor Stall Torque)
P(Cable Pull) = 89,569 lb (Limit Torque Coupling Should Slip Before This Condition is Reached)
For 70/30 Split, P = 62,698 lb
$W = 2{,}700$ lb, $W_1 = 875$ lb
$P_x = P \sin 2^\circ = 2{,}188$ lb
$P_y = P \cos 2^\circ = 62{,}660$ lb
$T_F = P \sin 1^\circ = 1094$ lb
$T_{FAX} = - T_{FBX} = (1094 \times 15)/28 = 586$ lb

Hoist Drum Shaft

Shaft Material HRS C-4150 ASTM A-434 Grade B, C Heat Treated, Yield Point = 80,000 psi, Tensile Strength = 103,000 psi. Per ASME Code Allowable Shear Stress = 0.3 x YP or 0.18 x TS. Use lesser of the two. Keyway, Fillet Factor = 0.75 x Allowable Shear Stress. Maximum Unit Stress is ≤ 0.75 x Allowable Shear Stress.

80,000 x .3 = 24,000 psi Yield Point
103,000 x .18 = 18,540 psi Tensile Strength
0.75 x 18,540 = 13,905 psi

$$d^3 = \frac{16}{\pi x \tau} \sqrt{(K_m M)^2 + (K_t T)^2}$$

$$\tau = \frac{16}{\pi x d^3} \sqrt{(K_m M)^2 + (K_t T)^2}$$

Machinery Handbook 21st ed.

where τ = shear stress

- d = shaft diameter
- K_m = 1.5 for gradually applied or steady loads
- K_m = 1.5-2.0 for suddenly applied loads, minor shock
- K_t = 1.0 for gradually applied or steady loads
- K_t = 1.0-1.5 for suddenly applied loads, minor shock
- Use K_m = 1.5, K_t = 1.25

Shaft size (minimum) based on previous calculation for torsional deflection = 6.8 in. Use a nominal 7.0 inch shaft.

Check Stress at M_{max} = 103,425 lb-in, located at P. The shaft OD at this location = 7.191"(sized for an interference fit with drum hub ID). And check stress at shoulder location where shaft diameter = 6 15/16" and M = 45,641 lb-in. The calculations for M_{max} and M are not shown in this example.

Normal Operating Condition - Full Load Motor Torque

Full Load Motor Torque - 70/30 Split

T_n = 0.94 x 564.5 x 796 x 0.7 = 295,600 lb-in

$$\tau = \frac{16}{\pi x 7.191^3} \sqrt{(1.5x103{,}425)^2 + (1.25x295{,}600)^2} = 5488 psi$$

$$\tau = \frac{16}{\pi x 6.9375^3} \sqrt{(1.5x45{,}641)^2 + (1.25x295{,}600)^2} = 5732 psi$$

Normal Operating Condition - Brake Torque

Brake Torque - 50/50 Split

T_b = 0.94 x 564.5 x 100 x 12 x 0.5 = 318,400 lb-in

$$\tau = \frac{16}{\pi x 7.191^3}\sqrt{(1.5x103{,}425)^2 + (1.25x318{,}400)^2} = 5850psi$$

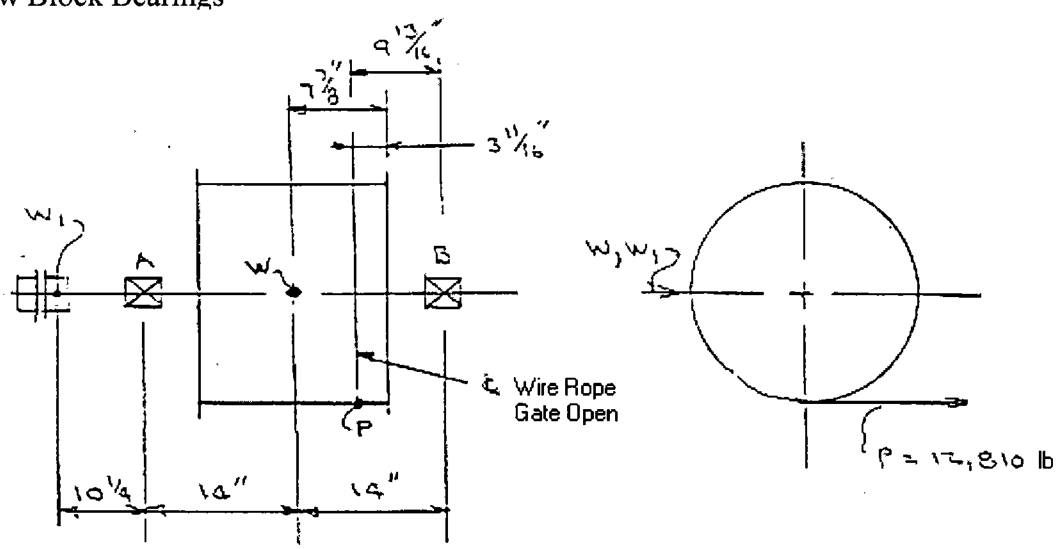
$$\tau = \frac{16}{\pi x 6.9375^3}\sqrt{(1.5x45{,}641)^2 + (1.25x318{,}400)^2} = 6158psi$$

Normal Operating Condition - Motor Stall Torque
Max. Motor Torque – 50/70 split
T_m = 0.94 x 564.5 x 796 x 3.25 x 0.7 = 960918 lb-in
Use K_m = 1.5, K_t = 1.0 for this condition

$$\tau = \frac{16}{\pi x 7.191^3}\sqrt{(1.5x103{,}425)^2 + (1.0x960918)^2} = 13{,}323psi$$

$$\tau = \frac{16}{\pi x 6.9375^3}\sqrt{(1.5x45{,}641)^2 + (1.0x960{,}918)^2} = 14{,}687psi$$

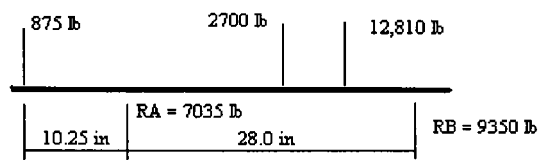
Split Pillow Block Bearings



$W_1 = 875$ lb, W = 2,700 lb

$$R_A = \frac{9.813x12{,}810 + 14x2700 + 38.25x875}{28} = 7035 \text{ lb}$$

$R_B = (12{,}810 + 2{,}700 + 875) - 7{,}035 = 9{,}350$ lb



R = Maximum Bearing Load (Normal Operating Condition) = 9,350 lb

This Load Occurs At Bearing B When Gate Is Fully Open.
The Bearing Size Is Dictated By The Required Shaft Size. From Manufacturer's Catalog Data Select a Split Pillow Block, Self-Aligning Spherical Roller Bearing For a 6 15/16" Diameter Shaft.

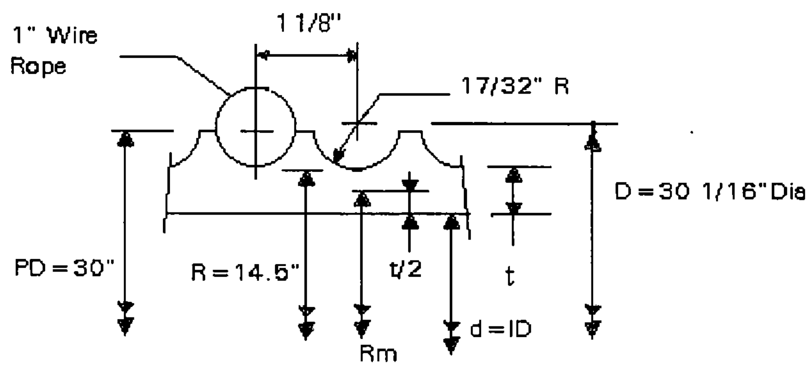
Calculate the Required Radial Rating (no Thrust Load)

$$RRR = \frac{RxLFxAF}{SF} = 18{,}300 \text{ lb and } LF = \frac{RRRxSF}{RxAF}$$

Where, R = 9350 lb
LF = Life Factor = 2.6 (Based on 75,000 Hours B-10 Life)
AF = Application Factor = 1.5 (Conservative Estimate)
SF = Speed Factor = 2.0 (Conservative Estimate)
Typical Cataloged Load Ratings (RRR) for this Bearing Size & Type = 139,000 lb

$$LF = \frac{139{,}000x2.0}{9350x1.5} = 19.8$$

Hoist Drum



b = Rope Diameter. = 1"
l = Pitch = 1 1/8"
t = Thickness Under Rope
L = Bearing Centers = 28"
F = Wire Rope Pull
R_m = Mean Rim Radius
P_{cr} = Critical Pressure
P = Radial Pressure

For Bending

$\sigma_b = M/Z$, where Z is the section modulus

$Z = 0.098 \times (D^4 - d^4)/D$

Assume Load F acts at L/2 for maximum moment

$M = FL/4$

$\sigma_b = FL/4Z = 7F/Z$

For Crushing

$\sigma_c = F/lt$ (From Crane and Hoist Engineering by Shaw Box)

$P = 2F/(2R_m \times l)$, therefore $\sigma_c = PR_m/t$ or $P = \sigma_c\, t/R_m$

$P_{cr} = 8.24 \times 10^6 \left(\dfrac{t}{R_m}\right)^3$ From Strength of Materials Part II by Timoshenko

$\sigma_{cr} = P_{cr} R_m/t$

F_{normal} = 12,810 lb (70/30 Normal Load)
F_{peak} = 19,292 lb (70/30 Peak Load (Full Load Motor Torque))
$F_{stall\ 1}$ = 62,698 lb (70/30 Motor Stall)
$F_{stall\ 2}$ = 44,784 lb (50/50 Motor Stall)

$F_c = 2.1 \times F_{peak}$ (70/30) = 40,513 lb*
*F_c= Maximum Torque Slip Coupling Can Transmit Before It Slips - Based On Coupling Setting of 190% $\pm$ 20% Of Full Load Motor Torque.

Calculate P_{cr} and σ_{cr}

For t = 1.25", R_m = 13.875"
P_{cr} = 6,025 psi, σ_{cr} = 66,878 psi*

For t = 1.5", R_m = 13.75"
P_{cr} = 10,698 psi, σ_{cr} = 98,063 psi*

*These values exceed the Yield Point (Y.P.) of the material, therefore the Y.P. is the governing criteria. The table shows that t = 1.5" provides acceptable stress levels.

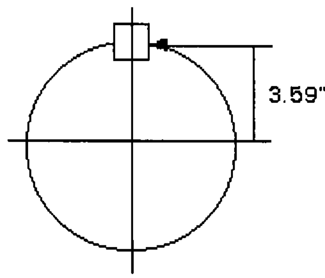
F (lb)	1 x t (in^2)	σ_c (psi)	t/R_m	P (psi)	σ_b (psi)	$\sigma_R = \sqrt{\sigma_c^2 + \sigma_b^2}$ (psi)
t = 1.25"						
12,810	1.406	9,111	0.090	820	123	9,112
19,292	↓	13,721	↓	1,235	187	13,723
62,698		44,593		4,013	607	44,597
44,784		31,852		2,867	433	31,855
40,513		28,814		2,593	392	28,817
t = 1.5"						
12,810	1.6875	7,591	0.109	827	106	7,592
19,292	↓	11,432	↓	1,246	160	11,434
62,698		37,154		4,050	519	37,158
44,784		26,539		2,893	371	26,541
40,513		24,008		2,617	335	24,010

For Hoist Drum Shell Use 30" OD X 26" ID
Steel Plate ASTM A 516 Grade 70.
Yield Point - 42,000 psi
Tensile Strength - 70,000 To 90,000 psi

The drum is composed of the shell, two end plates, a hub and the shaft. The end plates attach the shell to the hub. The hub, having an outside diameter of 11 inches, has an inside diameter sized to provide a class FN 2 fit to the shaft outside diameter. The shaft is also keyed to the hub with a 1.75" w x 1.5" d x 12" long key.

Key Design

Shaft Diameter = 7.191”
Try Key Size = 1.75” wide x 1.5” deep x 12” long



F=T/R, σ = F/A and then σ = T/RA
RA = 3.59 x (1.75 x 12) = 75.4 in^3
Rated Brake Torque (50/50 Split) = T_b = 318,400 lb-in
Maximum Motor Stall Torque (70/30 Split) = T_m = 960,900 lb-in

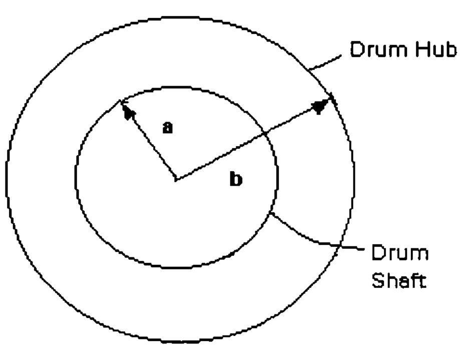
From the Distortion Energy Theory
$\sigma_{sy} = 0.577\sigma_{yp}$
Use C-1045 CFS Key Stock
Yield Point = 75,000 psi
Tensile Strength = 90,000 psi
σ_{sy} = 43,275 psi

Safety Factor (SF) = σ_{sy}/σ

σ_b = 318,400/75.4 = 4223 psi	SF = 43275/4223 = 10.2
σ_m = 960,900/75.4 = 12,744 psi	SF = 43275/12,744 = 3.4

Key Design is Adequate

Check Shaft and Hub Class FN 2 Interference Fit
Size Range = 7.09” to 7.88”



$$P_c = \frac{E\delta}{a}\left[\frac{(b^2 - a^2)(a^2)}{2a^2b^2}\right]$$

where P_c = Contact Pressure
δ = Interference, for Class FN 2 = -.0032" to -.0062"
a = 7.191/2 = 3.595"
b = 11.0/2 = 5.5"
E = Modulus of Elasticity = 30 x 10^6 psi

P_c (Minimum) = 7650 psi
P_c (Maximum) = 14,800 psi

Check Torque for Minimum Pressure
Drum length (L), considering the gate travel, and allowing three dead wraps (gate closed) = 15.75"
T = f ($2a^2$) (P_c) (π) (L) = 0.12 x 2 x 3.595^2 x 7650 x π x 15.75 = 1,174,000 lb-in (0.12 is a constant recommended by the crane industry)
Class FN 2 Fit is adequate

2. Sector Gate - Machinery Design

Typical calculations for determining closing pintle torque for a reverse head.

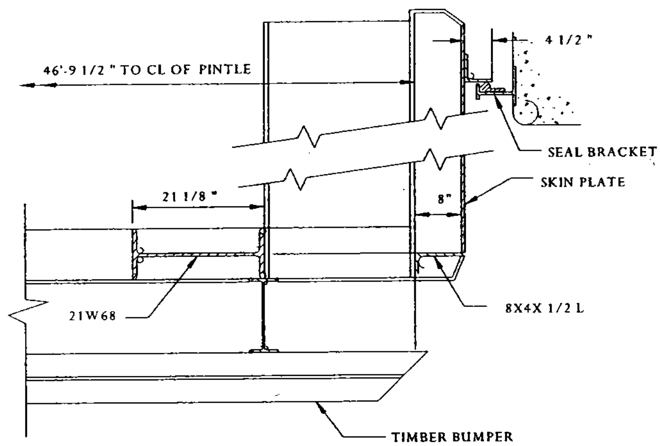
Closing torque is composed of

- Hydrodynamic forces acting on the nose of the gate,
- Hinge and pintle friction - composed of friction from gate's weight and hydrostatic head.

There is no torque due to seal friction because the reverse head lifts the bottom seal. There are no reverse head seals.

Design Conditions: 5' Reverse Head
16' Tailwater

a. Hydrodynamic Torque: To obtain peak pintle torque due to hydrodynamics see WES Report H-70-2 Appendix A, Plate A4, Figure a. Nose of the design gate is as shown below.



NOSE AND SEAL DETAIL
OF THE DESIGN GATE

Projected width of miter beam, skin plate rib and
seal plate bracket of design gate= 21.125" +8" +4.5"= 33.625"

For the design conditions Figure a indicates a hydrodynamic
torque = 200 Ft-Kips.

Figure a was developed for a gate with a radius of 42 feet and total projected width of miter beam, skin plate rib, and seal bracket of 30.375" (see WES Technical Report, H-70-2, Appendix A, Plate A1. The design gate has a gate radius of 46.792' and a projected width of miter beam, skin plate and seal plate bracket of 33.625". To obtain the hydrodynamic torque for the design gate it is necessary to apply correction factors for the gate radius and projected width as follows:

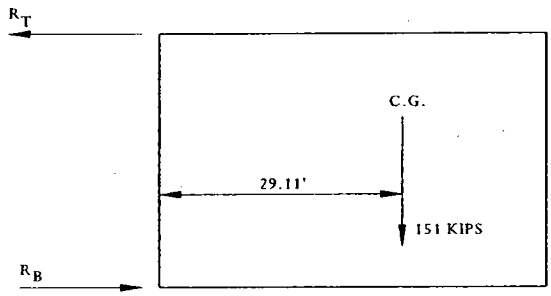
Hydrodynamic torque for the design gate =
200 Ft-Kips(33.625"/30.375") (46.792'/42') = <u>247 Ft-Kips</u>

b. Hinge and Pintle Friction:
Design Data:
Spherical Hinge Diameter = 12"
Spherical Pintle Diameter = 18"
Gate Weight = 151 kips @ cg 29.11' from vertical hinge & pintle centerline.

Assume:

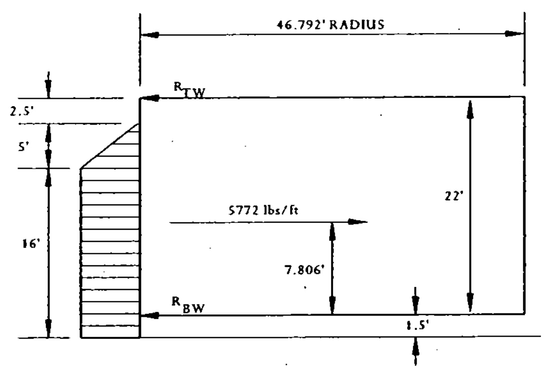
- Vertical dead weight reaction acts at one half of the spherical pintle radius or 4.5".
- Coefficient of friction for bushings = 0.25.
- Static water load and horizontal dead weight reactions act at the hinge and pintle radius

Hinge and pintle load from gate's weight:



$R_T = R_B = 151$ Kips (29.11') / 22' = 200 Kips

Hinge and pintle load from the hydrostatic head:



53.716' (.5(62.4 lbs/ft^3) (5')^2 + 62.4 lbs/ft^3(5')(16')) = 310.05 Kips

Where 53.716' is the cord length of the skin plate.

Centroid of net static water load is 7.806' up from the pintle, therefore,
Pintle reaction = R_{BW} = 310.5 (22'-7.806')/(22') = 200.33 Kips and

Hinge reaction = R_{TW} = 310.05 Kips - 200.33 Kips = 109.72 Kips.

Net pintle horizontal reaction = -200.33 Kips + 200 Kips = 0.33 Kips

Net hinge horizontal reaction = 200 Kips + 109.72 Kips = 309.72 Kips
Friction from horizontal reactions
= 0.25 (9/12)' (.33 Kips) + 0.25 (6/12)' (309.72 Kips) = <u>38.78 Ft-Kips</u>

Friction from vertical reaction = 0.25 (4.5/12)' (151 Kips) = <u>14.16 Ft-Kips</u>

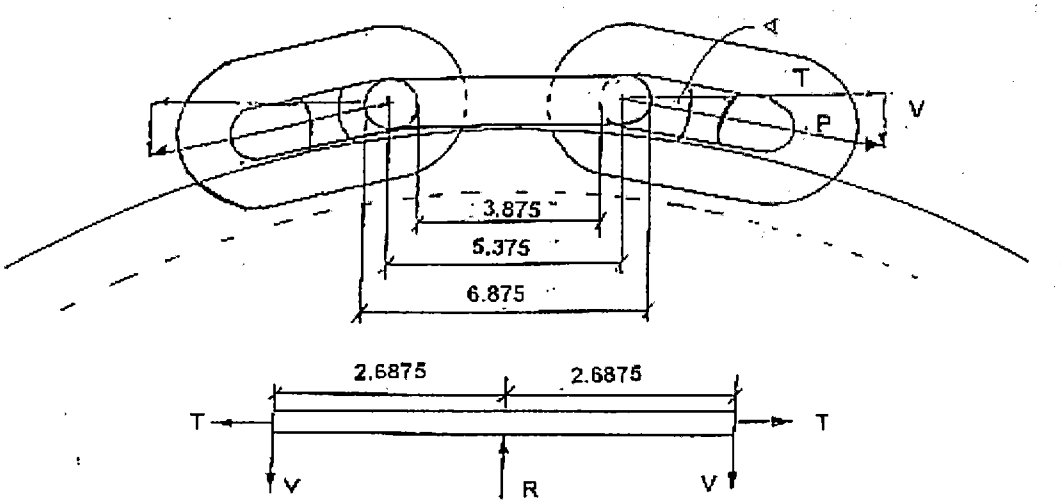
Total calculated torque = 247 Ft-Kips + 38.78 Ft-Kips + 14.16 Ft-Kips = <u>299.94 Ft-Kips</u>

Applying a service factor of 1.5, the design operating torque =
1.5 (299.94 Ft-Kips) = **<u>449.9 Ft-Kips</u>**

3. Round Link Chain - Grooved Drum and Pocket Wheel

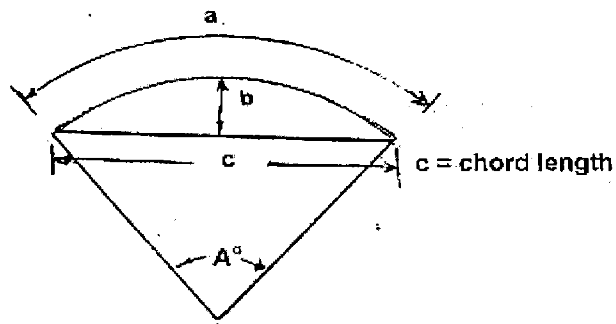
a. Chain Link Bending Stresses around Grooved Drum. Assume 1 ½ inch diameter round link with 3.875 inch pitch over a 41.69 inch diameter grooved drum.

Design load = 45,000 lb. x 275% overload
= 126,000 lb.



Material ASTM - A391, Fy = 160,000 psi, fy = 144,000 psi.

combined $\frac{fa}{Fa} + \frac{fb}{Fb} \leq 1$ (AISC)

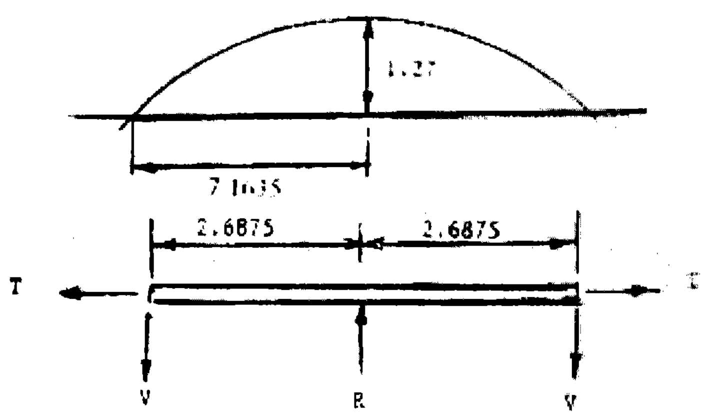


a = 3.875 x 3 (links) + 1.5 x 2 = 14.625 in.

$A° = 57.29578 \times \frac{14.625}{20.845} = 40.2°$

$c = 2 \times 20.845 \sin \frac{40.2}{2} = 14.327$ in.

$b = \frac{14.327}{2} \tan \frac{40.2}{4} = 1.27$ in



$\tan \sphericalangle = \frac{1.27}{7.1635} = 0.1773 \therefore \sphericalangle = 10.05°$

T = P cos 10.05° = (126,000) (.9846) = 124,065 lb.
V = P sin 10.05° = (126,000) (.1745) = 22,000 lb.

$A = 2\,(1.5)^2\,(\pi/4) = 3.53$ sq in.

$$S = \frac{(\pi)\,(1.5)^3\,(2)}{32} = 0.663 \text{ cu in.}$$

T = 124,065/3.53 = 35,145 lb/sq in.

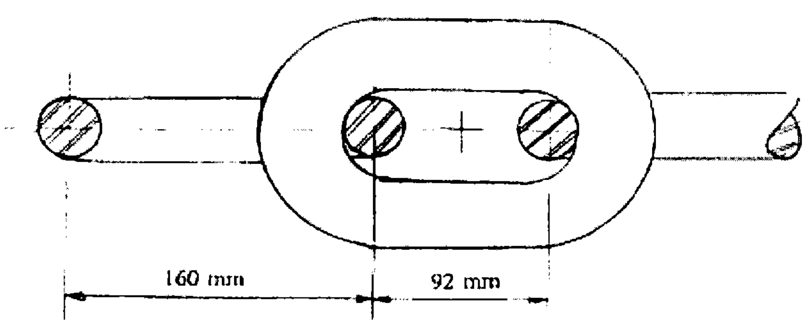
$$M = 2\,\frac{(22000)\ (5.375)}{4} = 59{,}125 \text{ lb in.}$$

$$v = \frac{M}{S} = \frac{59{,}125}{0.663} = 89{,}178 \text{ lb/sq in}$$

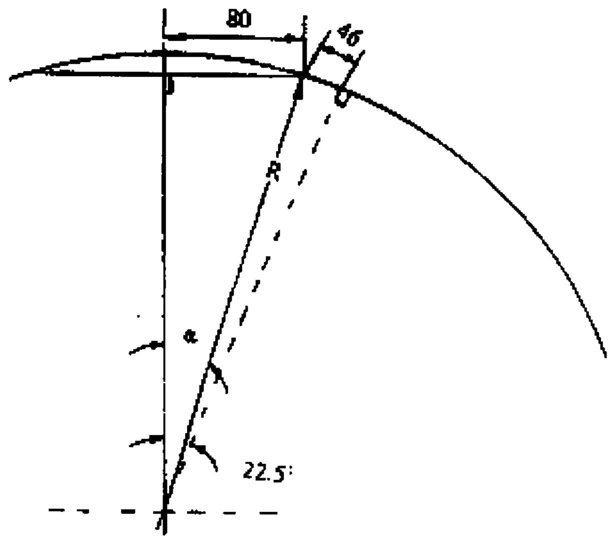
$$v \text{ combined} = \frac{35145}{144{,}000} + \frac{59{,}125}{144{,}000} = 0.65 < 1 \text{ (o.k.)}$$

b. Determine Pocket Wheel Pitch Diameter

(1) Based on Chain geometry
34 x 126 chain (DIN 22252)



Reference to shop drawing (Columbus McKinnon)

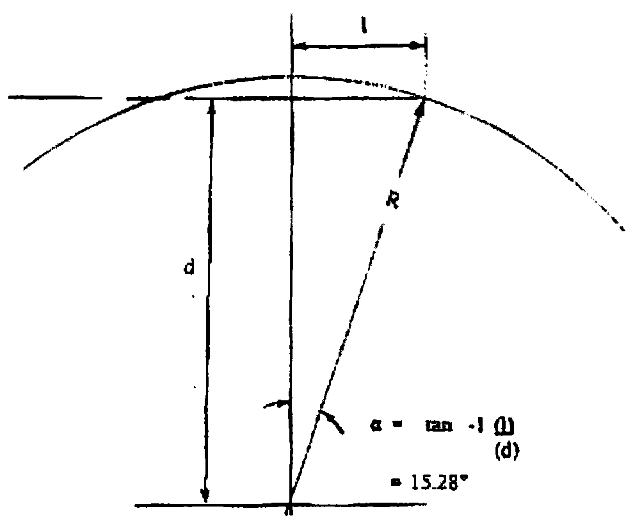


$$R = \frac{80}{\sin \alpha} = \frac{46}{\sin (22.5 - \alpha)}$$ solving for $\alpha = 14.32°$ and $R = 323.39$ mm

$$PD = 2R = 2 \times \frac{323.39}{25.4} = 25.46 \text{ in.}$$ (Pitch diameter based on chain geometry)

(2) Based on Pocket Wheel Geometry

Reference to shop drawing 8-pocket sprocket (Columbus McKinnon)



$$d = \begin{pmatrix}\text{distance from Cutter face} \\ \text{to C.L. of Pocket Wheel bore}\end{pmatrix} + \begin{pmatrix}\text{radius of} \\ \text{chain stock}\end{pmatrix}$$

$$d = 11.54 + \frac{34\text{mm}}{2 \times 25.4} = 12.2093 \text{ in.}$$

$$l = \begin{pmatrix}\text{center of pocket} \\ \text{to center of cutter}\end{pmatrix} + \begin{pmatrix}\text{radius of} \\ \text{cutter}\end{pmatrix} - \begin{pmatrix}\text{radius of} \\ \text{chain stock}\end{pmatrix}$$

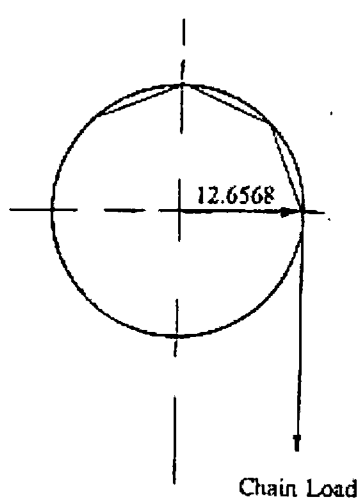
$$l = 1.845 + \frac{4.32}{2} - \frac{34 \text{ mm}}{2 \times 25.4} = 3.3357 \text{ in.}$$

$$R = \sqrt{(12.2093^2 + 3.3357^2)} = 12.6568 \text{ in.}$$

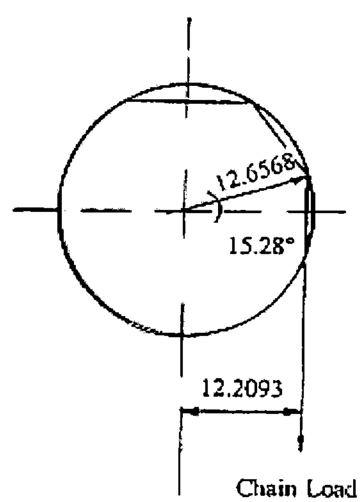
PD = 2R = 25.31 in. (Pitch diameter based on pocket wheel geometry)

(3) Pocket Wheel Pitch Diameter

The difference between the pitch diameter based on chain geometry and that based on the pocket wheel geometry is an indication of the slop desired to permit free engagement and disengagement of the chain links to the wheel pockets as the wheel rotates. The radius at which the load acts varies as the wheel rotates from a maximum of 12.6568 to a minimum of 12.2093, as determined by the computation based on pocket wheel geometry.



Position of pocket wheel
a max. radius



Position of pocket wheel
a min radius (15.28° rotation)

For purposes of determining speed of rotation and torque requirements for hoist equipment, a radius of 12.6568 will be used. This gives a pitch diameter of 25.3 in.

c. Presentation of Hoist Capacity

$$\underline{\text{Pocket Wheel r.p.m.}} = \frac{\text{hoist speed x 12 in/ft.}}{\pi \text{ x pocket wheel dia-in}} \quad \text{(Equation 1)}$$

$$\underline{\text{Gear train reduction}} = \frac{\text{motor speed - rpm}}{\text{pocket wheel - rpm}} \quad \text{(Equation 2)}$$

(1) Design Conditions

	MOVING		STALLED
	Normal	Peak	Max
Total load on chains:	120k (assumed)	136k	336k
Per side of gate:	60k(1)	68k	168k
Factor of Safety:	5(1)	3	-
Maximum unit stress:	-	-	75% yield
% Motor rated torque:	100%	≤ 115% Continuous	≤ 280%
Nominal hoist speed:	1.0 FPM	< 1.0 FPM	

Reference: EM 1110-2-2702

(2) Power Equation:

$$HP = \frac{vL}{33 \text{ x } \eta}$$

where: v = Chain speed (FPM)
L = Load (LBS)
η = efficiency of powertrain

η (powertrain) = η (triple box) x η (double box) x η (open gearing, 1 reduction)

Open gearing consists of the following:

- chain/pocket wheel efficiency 0.90 assumed
- spur gear set 0.97
- two sets of antifriction bearings @ 0.98 ea 0.98^2

Efficiencies quoted by gearbox manufacturers are typically high, therefore, use the following:

η (triple box) = 0.90
η (double box) = 0.95

η (powertrain) = $0.90 \times 0.95 \times 0.90 \times 0.97 \times 0.98^2 = 0.7169$

Use 0.72

(3) Sample Calculations

$$\text{Solve for HP} = \frac{1.0 \times 120}{33 \times 0.72} = 5.05 \text{ say } \underline{5 \text{ HP}}$$

$$\text{and } v = \frac{5 \text{ HP} \times 33 \times 0.72}{120} = .99 \text{ say } \underline{1 \text{ FPM}}$$

d. Chain Locker Dimensions.

V_L required = Required volume of chain locker

where V_L required = $0.85\ d^2\ L$

and d = chain link diameter - in
L = chain length /6 - fathoms

Sample calculation:

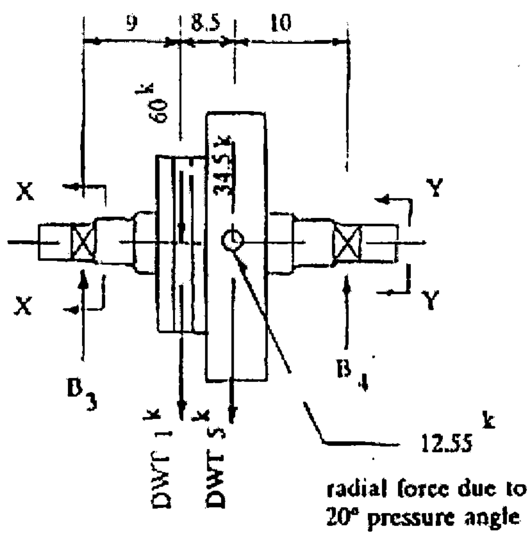
Assume 1½ in. diameter chain, 46 ft. in length

$$V_L \text{ required} = 0.85\ (1.5)^2\left(\frac{46}{6}\right) = 14.7 \text{ cu. ft.}$$

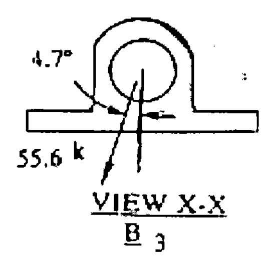
The following tabulation showing chain locker depth for various selected diameters is helpful in determining the desired locker size, based on machinery locations and space limitations.

Dia-in.	Depth-in.
12	225
18	100
24	56
30	36

e. Bearing Selection - Pocket Wheel Shaft Load Computation



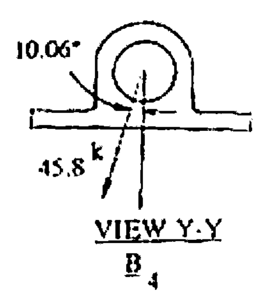
Check for normal load based on 60k chain load and FOS. = 5



$$\text{B3 vertical force} = \frac{(61)\,(18.5) + (39.5)\,(10)}{27.5} = 55.4\text{k}$$

$$\text{B3 horizontal force} = \frac{(12.55)\,(10)}{27.5} = 4.55\text{k}$$

$$\text{Resultant force} = 55.6\text{k} \ @ \ \tan^{-1}\frac{(4.55)}{(55.4)} = 4.7^\circ$$



$$\text{B4 vertical force} = \frac{(39.5)\,(17.5) + (61)\,(9)}{27.5} = 45.1\text{k}$$

B4 horizontal force = $\frac{(12.55)(17.5)}{27.5} = 8k$

Resultant force = 45.8k @ $\tan^{-1}\frac{(8)}{(45.1)} = 10.06°$

(1) Similar computations should be made for chain loads based on a unit stress not in excess of 75% of the yield point of the material as called for in EM 1110-2-2702. Bearing sizing philosophy will be to specify for the highest resultant force as the minimum static load capacity required.

(2) The bearings should have a life expectancy requirement of 10,000 hours B-10 life with loads assumed equal to 75% of the maximum load.

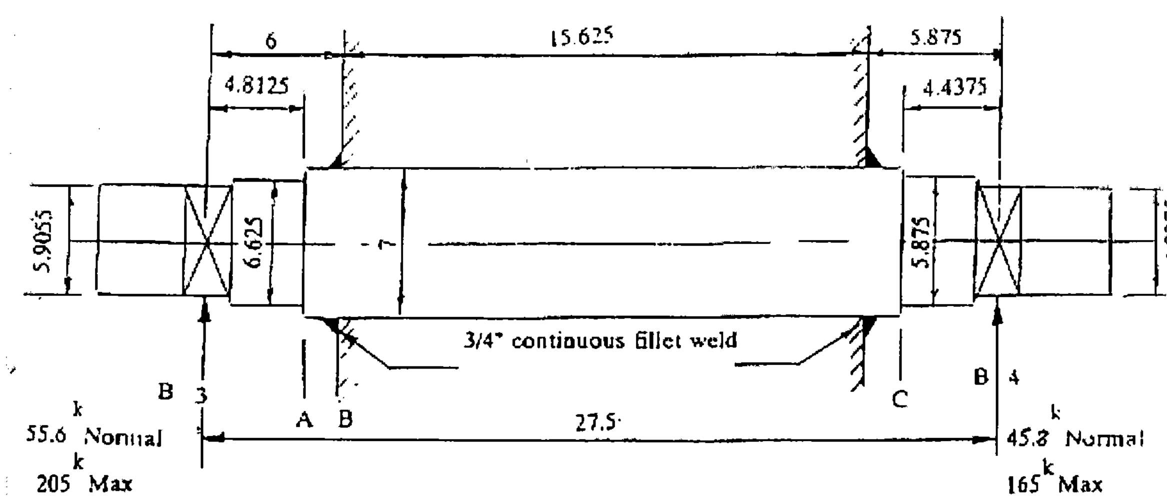
(3) End unit bearings may be subjected to a range of loads corresponding to the following:

a. Even-split of NORMAL rated load to each end unit.

b. Even-split of motor stall torque (½) (2.8 x normal rated load) to each end unit.

c. Maximum UNEVEN - split of motor stall torque to each end unit.

f. Pocket Wheel Shaft - Stresses and Material Selection.



(1) Compute Normal stresses at sections 'A', 'B' & 'C'.

	Section A	Section B	Section C
Area	34.5 in^2	38.5 in^2	27.1 in^2
Section Modulus ($\pi d^3/32$)	28.5 in^3	33.7 in^3	19.9 in^3
Normal Bending Moment	267.6 kp-in	333.6 kp-in	203.2 kp-in
Bending Stress (M/Z)	9.4 ksi	9.9 ksi	10.2 ksi
Shear Stress (V/A)	1.6 ksi	1.4 ksi	1.7 ksi

Combine stresses for 'C'

$$\tau_{max} = \sqrt{\left(\frac{10.2}{2}\right)^2 + (1.7)^2} = 5.38\ ksi$$

$$\sigma = \left(\frac{10.2}{2}\right) + 5.38 = 10.5\ ksi$$

(2) Compute Maximum stresses at sections 'A', 'B' & 'C'.

	Section A	Section B	Section C
Bending	$(9.4)\frac{(205)}{(55.6)} = 34.7$ ksi	$(9.9)\frac{(205)}{(55.6)} = 36.5$ ksi	$(11.4)\frac{(165)}{(45.8)} = 36.8$ ksi
Shear	$(1.6)\frac{(205)}{(55.6)} = 5.9$ ksi	$(1.4)\frac{(205)}{(55.6)} = 5.2$ ksi	$(1.7)\frac{(165)}{(45.8)} = 6.1$ ksi

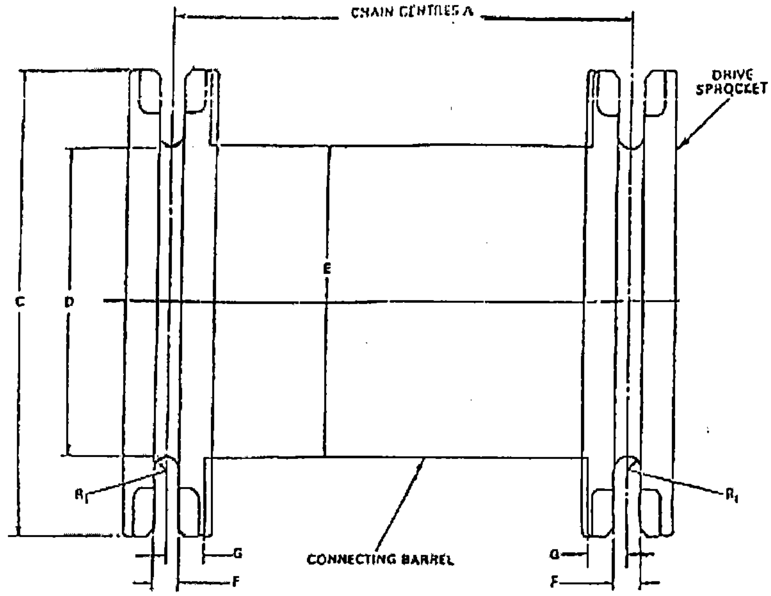
Combine stresses for 'C'

τ max = 19.4 ksi; σ = 37.8 ksi

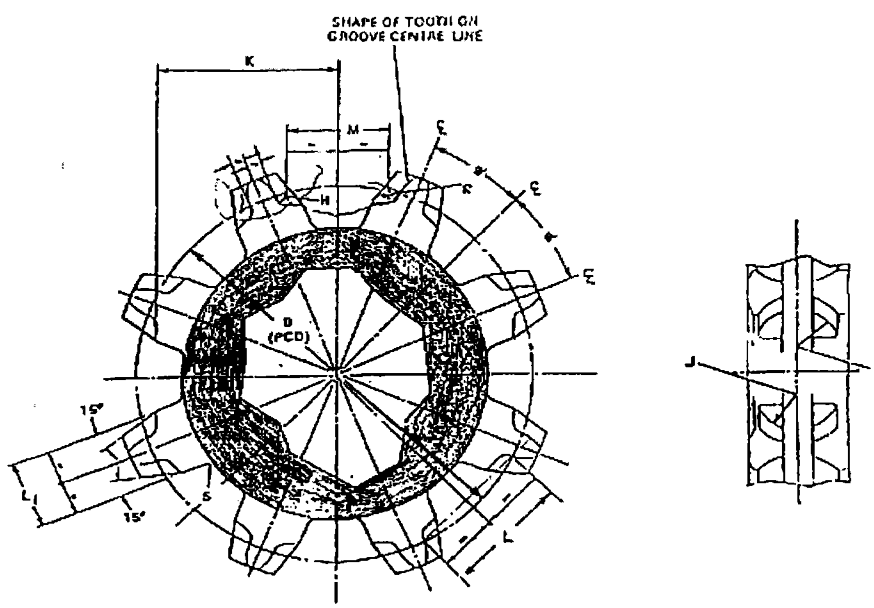
Use: ASTM A-668 Class E (UTS = 85 ksi, Fy = 44 ksi)
with supplementary requirement S4 (carbon content for welding)

F.O.S. = $\frac{85}{10.5} = 8.1$ and Fy = $\frac{37.8}{44} = 86\%$

g. Sprocket Assembly - Design Formula



Typical Sprocket Assembly



Sprocket Ring Profile

(1) Dimension B - Pitch circle diameter (theoretical)

$$B = \sqrt{\frac{P^2}{\sin^2\left(\frac{\theta}{2}\right)} + \frac{d^2}{\cos^2\left(\frac{\theta}{2}\right)}}$$

where:

d = Nominal diameter of chain link material.

P = Nominal pitch of chain link.

$\theta = \frac{360}{2N}$ degrees

N = Number of teeth in sprocket

The value for B obtained to be taken to nearest lower whole number.

(2) Dimension C - Overall diameter (reference)
C = B + 2d

NOTE - Actual diameter to be agreed between purchaser and manufacturer.

(3) Dimension D - Groove diameter
D = Diameter under vertical chain links minus a diametral clearance.

NOTE: Actual diameter to be agreed between purchaser and manufacturer.

(4) Dimension E - Barrel diameter
E = 2K + d – 2 (Bolt center line to bottom of scraper bar) – 5

(5) Dimension F - Sprocket groove width
F = 1.25d

(6) Dimension G - Groove center line to inside face of sprocket recess

$G = b_t - (0.5e + 0.5V_u + 3.5)$

where:

e = Diameter of nut across corners.

V_u = Clearance between bolt and hole of shackle connectors.

b_t = Chain center to hole center of shackle connector.

Dimension to be maintained in vicinity of nut and bolt only.

(7) Dimension H - Root radius
$H = 0.5d$

(8) Dimension J - Pocket plan radius (nominal)
J = Maximum outer radius of shackle connector and is measured on a line $K + 0.5d$ from sprocket center line.

NOTE: If a working clearance is required it should be agreed to between purchaser and manufacturer.

(9) Dimension K - Height from sprocket center to bottom of the pocket

$$K = 0.5\left[\frac{P}{\tan\left(\frac{\theta}{2}\right)} - d\tan\left(\frac{\theta}{2}\right)\right] - 0.5d$$

The values for K obtained to be taken to nearest half millimeter.

(10) Dimension L - Length of pocket
$L = 1.075\ P + 2d$

(11) Dimension M - Pocket centers (reference)
$M = 1.075\ P + d$

(12) Dimension R - Tooth flank radius (reference)
$R = P - 1.5d$

Radius to be struck from a line which is K + 0.5d from sprocket center line.

(13) Dimension R1 - Groove radius
R1 = 0.5d

(14) Dimension S - Radius at root of tooth stub
S = 0.5d

h. Typical Hoist Arrangement - Summary of Service Factors